EXPERIMENTAL AND NUMERICAL STUDY OF VIBRATION MONITORING APPLIED ON GEAR TRANSMISSION SYSTEMS

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Abstract: Predictive maintenance asks for efficient indicators and methods to detect incipient failures. Numerous studies have been made in this domain, but some difficulties stay to isolate the defected elements, to foresee through signal analysis, causes that produce faults, especially when differe technological elements are involved simultaneously (gears, bearings, ...). In the first part of this paper, experimental study have been performed on gear faults: manufacturing faults (profile error, deviation, ...) and operating faults (wear, pitting, ...) in order to establish a kind of data base for fault identification. Then, operating faults were examined and a procedure of identification of faults nature have been developped. Statistical treatement of both time and time-frequency (wavelet analysis) domains give arise to robust indicators. The second part deals with the definition of thresholds for machinery stops. Calculations of the dynamic behaviour of the geared system show that how a gear fault modify the gear profile and the kinematic transission error, while stiffness modification have almost no significant effect on the system response.

Key Words: Diagnostics; condition monitoring; maintenance; vibration analysis, signal processing; numerical simulation; gears

INTRODUCTION: This study takes place in a research program dealing with the condition monitoring of gearboxes, and particularly the analysis of the vibration signatures of spur and helical gear faults [1]. Machine monitoring using vibration measurements is rapidly established itself as a worthwhile techniques in modern industry. The analysis permit to characterize results of faults and gear damages [2, 3, 4]. The research program includes several steps: the study of design and manufacturing parameters as the tooth profile modification, misalignement, pitch module, center axe distance, ... This part enable users to optimize gear design parameters and the definition of the reference state of the studied system. The second step is the study of effects produced by local and distributed faults on spur and helical gears. The aim is to isolate significant parameters from the system response which indicate the presence and the nature of faults [5] and then establish warning level for fault progressing. In order to accomplish this task, experimental and numerical simulation have been undertaken in the same time. In this paper, only faults on gears will be considered. Vibration from bearing housing seems to be the suitable system response to be used. The working method is as follow:

- making tests for gears without fault,

- create fault on one gear, then reproducing the same set of tests,
- processing the acceleration signal from bearing housing in different domains (time, frequency and time-frequency),
- extracting the influence of signal parameters through a statistical design procedures.

Measurement were carried out for several operating conditions (speeds and loads), descriptors of bearing acceleration levels were compared with and without faults. Finally, and in order to define warning levels, the studied system is modelled using the Finite Elements Method [6]. The aim is to correlate the calculated dynamic load evolution with respect to the gear fault dimension, and indicator evolution issued from experimental measurements.

EXPERIMENTAL PROCEDURE

TEST RIG: The test stand (Fig. 1) is constituted of two shafts of 60 mm in diameter, mounted on two rolling bearings and coupled with gears. Test gears were clamped with nuts at the operating end of each shaft and centred by involute splines so as to limit variations of eccentricity. After gear clamping, a special nose was fixed at each shaft end, in order to measure their angular positions during motion with optical encoders (C1-C2), used for transmission error measurement [7]. The output shaft (1) and its rolling bearings were fixed on a special mounting, made of thick intermediate plates. It permits to impose small misalignments. The input shaft (2) was driven by a 120 kW DC motor. The output shaft is braked by a DC motor. Rotating speed was varied between 0 and 6000 r/min and was feedback controlled. The input torque was varied independently from 0 to 150 Nm. The active part of the apparatus was fixed on a 7 tons rigid frame (3). Vertical and horizontal bearing accelerations have been measured with piezoelectric accelerometers A1 and A2 respectively. Other parameters (speed, torque, oil temperature) were also recorded for each test.

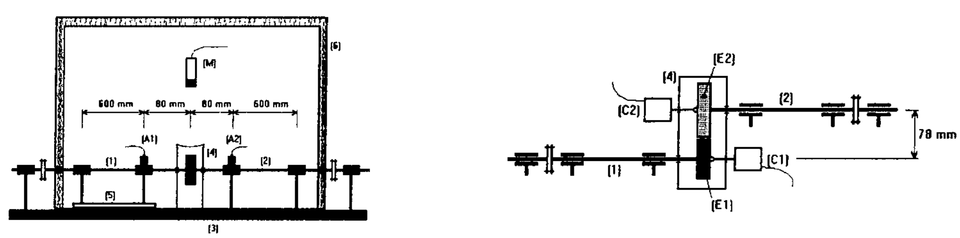


Fig. 1: Schematic arrangement of the test gear apparatus

SIGNAL ANALYSIS: Stored bearing acceleration signals have been processed using the asynchronous FFT. The spectrum is obtained from an average of 16 successive FFT signal acquisitions of 1024 points. Note that for this analysis, the amplitude of the first five harmonics of gear meshing frequency (H1, H2,..H5), the sum of the first five harmonics and the root mean square value (RMS) of the signal are considered. Beside time domain waveform study and spectral analysis, Fast Wavelet Transform analysis (FWT) has been used. Details and development of this technique are defined elsewhere [8]. This is a projection technique similar to the FFT. However, rather that simply decomposing the signal into sinusoids of varying frequencies, the data is represented as a projection onto a special function. This means that data set can be represented as time translations of a mother wavelet (a basis function) and as time dilatations (i.e. expanding the time scale of observation). Then, amplitude of the FWT coefficients can be visualized versus time (Fig. 2). The x-axis represents the time (ms), while the

y-axis represents octave bands of frequency (0-48 Hz, 48-97 Hz, etc ...). The FWT technique can be interpreted as octave band filters.

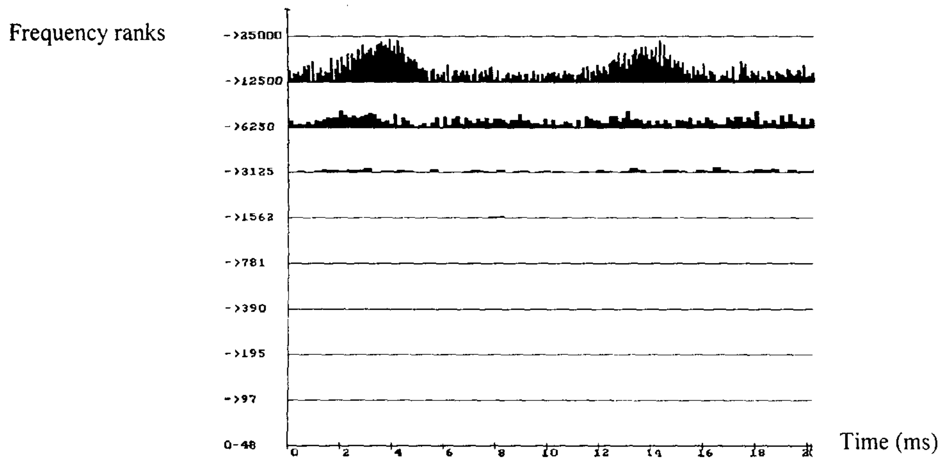


Fig. 2 : Wavelets representation. Spur gear.Distributed fault (12 daNm, 6000 rpm)

The statistical analysis has been also considered in order to quantify the effect of gear faults. Thus, for time domain and for wavelet coefficient analysis, the parameters are: Peak value, Peak to peak value, Standard deviation (s), Skewness, Kurtosis, Root Mean Square value (RMS) and Crest factor (peak/ RMS).

EXPERIMENTAL RESULTS: In a first time, the influence of design and manufacturing parameters on the dynamic behaviour of spur and helical gears (AGMA 12 quality and 51/52 teeth) was studied. The considered parameters were: deviation of 4', inclination of 4', tooth profile modification, center axis distance (78.02, 77.89 mm), module (2.1, 1.52 mm), contact ratio (1.3, 2, 3) and overlap ratio (0, 1, 2). Measurements have been done for several speeds (1000, 2000, 3500, 4750 and 6000 rpm) and several loads (0, 3, 6, 9 and 12 daNm). The conclusions of this part are summarized in Table 1.

Studied parameters	**Effects on acceleration**
Torque	For the harmonics H1 & H2 & for the sum of the first five harmonics, the influence is important, it is not so for the RMS.
Speed	The RMS is very sensitive for the different harmonics.
Contact ratio	The level of the different descriptors decreases with an increasing of contact ratio.
Deviation	Give arise to an increase of different descriptor levels.
Inclination	Does not bring any change in the level of the different descriptors.
Tooth correction	Give arise to a decrease of different descriptor levels.
Center axis distance	Important increase of the different descriptors has been remarked.
Pitch module	An increase of the module gives arise to a decrease of the level of different descriptors.

Table 1: Effects of design parametres on bearing acceleration

Local fault: The operating conditions are five speeds and five loads as above. For each test, the statistical parameters were calculated and the well-known Taguchi experimental design procedure [9] have been applied. The purpose of this analysis consists of first, defining the test to be realized, and second applying variance analysis procedure which enables us to extract the influence of parameters from the whole set of parameters being studied. The characteristics of the test gears are presented in Table 2.

characteristics	**spur gear**	**helical gear**
number of teeth	36/38	36/38
contact ratio ε_α	1.3	1.3
overlap ratio ε_β	0.0	1.0
helix angle	0.0	21°
face width of the tooth (b)	17.5 mm	17.5 mm
tooth thickness (e)	3.31 mm	3.31 mm
total depth of the tooth (h)	4.96 mm	9.30 mm
pitch module	2.0 mm	2.0 mm

T-ble 2: Characteristics of test gears

A local fault has been created along the pitch line of one tooth . It had a width of 0.863 mm and a 0.169 mm depth (Fig. 3). Tests have been made and signals obtained for spur gear with and without local fault have been processed. Beside the increase in amplitude of time domain waveform and of spectrum, due to fault, an increase of the FWT coefficients is also noted. This increase exists always in the last two octaves considered in the visualization.

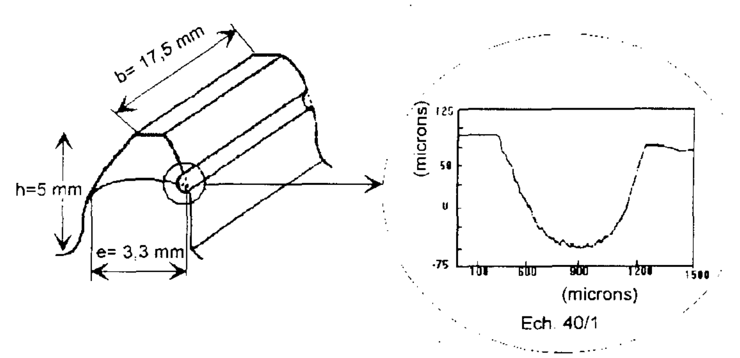


Fig. 3: Visualization of local fault

Statistical parameters of the three treatments have been calculated. Noting here that, in the case of the FWT analysis, only the last two octaves will be considered. A Taguchi experimental design procedure has permitted to extract the parameters that have been influenced by this type of fault. The final results of the Taguchi procedure is expressed in the Table 3. In the first column, the studied factors (fault, torque and speed) are noted. The other columns contain the statistical parameters with their influences in percents. Fault has an influence of more than 80 % on four of the seven studied parameters.

studied factors	parameters						
	Mean	Peak	RMS	s	skew	kurt.	crest factor
Fault	87 %	83 %	85 %	86 %	17 %	14 %	19 %
Speed	9 %	6 %	6 %	8 %	16 %	16 %	7 %
Torque	1 %	4 %	4 %	2 %	3 %	0 %	2 %
Residual	3 %	7 %	5 %	4 %	64 %	70 %	62 %

Table 3 : Influence of different statistical parameters

This procedure is applied for the two other signals. In summary, the parameters which have the largest influence are for :

- Time domain: peak value, peak to peak value, standard deviation and RMS,.
- Signal spectrum : H1, H2, $\sum_{i=1}^{5} H_i$ and RMS,
- Wavelet coefficients: peak value, mean, standard deviation and RMS.

Concerning the helical gear, the shape of the created local fault was similar to that of spur gear. This fault was of 0.859 mm width and its depth was of 0.033 mm. Tests have been performed and the statistical parameters have been calculated for the three treatments. The Taguchi statistical design procedure have been also applied. The same significant parameters as those considered in the case of spur gear tests have been found.

Uniformly distributed fault : In the case of local faults in spur and helical gears, good results have been obtained concerning the detection of incipient failure by statistical parameters, and particularly those of FWT coefficients. Now, the sensibility of these parameters in detecting a uniformly distributed fault will be considered. This fault has been induced on a helical gear of the same characteristics as the previous local fault gear. Test was recorded after every one hour. The speed of rotation was 3500 rpm with a 90 Nm torque. Lubrication was removed approximately 40 hours after the start of the test and at this time a fault uniformly distributed has been induced by scuffing. The results issued from the treatment of the recorded signal show that during the early period of the tests, no change have been pointed out for all the three signal presentations and also in all of their statistical parameters. While after inducing of the fault, amplitudes of time domain waveform, spectrum and wavelet coefficients were largely modified. This fault gives a modulation in time domain waveform that appears clearly with Fast Wavelet Transform analysis. This phenomenon can be attributed to an increase of the eccentricity effect on gear noise and vibrations. In fact, definition of uniformly distributed fault gives a single image of the one tooth fault. However, it can be reasonably considered that scuffing differs from one gear tooth to an another tooth. Then, new circular pitch and rotations occur. Tables 4 and 5 show the statistical parameters of fault. Results obtained from spectrum and FWT analysis are practically the same. On these tables, one can find the statistical parameters values with their rate of change, in percents, with respect to a baseline. The test after one hour is considered as a baseline against which test parameter are compared. It can be observed that a minimum of 10 % difference appears between the two considered gear situations. It can be notted too, that with an uniformly distributed fault, gears out of service with about 10% of level increase, while local fault gives more than 50% of level increase and gears still almost available. That is why, we have developped a methodology to identify the nature of fault, based on the use of statistical tests,

applied on the last two frequency rangs of the FWT analysis: KHI2 test identifies distributed fault and extreme values test, local fault [5].

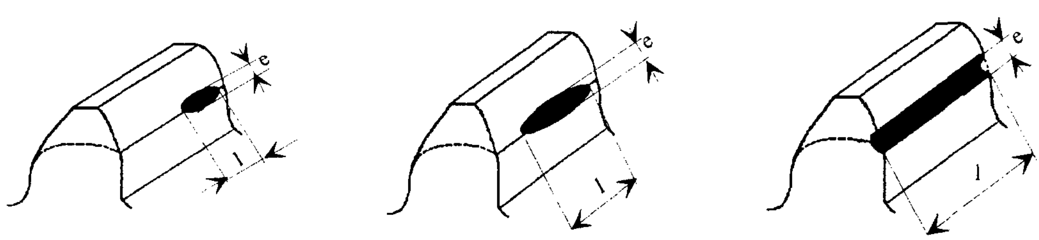
	statistical parameters			
	Peak (dB)	peak to peak (dB)	s (dB)	RMS (dB)
after one hour	66.1/ 0 %	72.1/ 0 %	57.2/ 0%	57.2/ 0 %
after 4 hours	69.3/ 5 %	75.1/ 4 %	61.4/ 7 %	61.4/ 7 %
after 8 hours	68.6/ 4 %	74.2/ 3 %	60.8/ 6 %	60.9/ 7 %
after 16 hours	69.8/ 6 %	75.3/ 4 %	61.7/ 8 %	62.0/ 8 %
after 32 hours	66.8/ 1 %	72.4/ 1 %	58.9/ 3 %	58.9/ 3 %
with fault	77.5/17%	82.9/15%	69.6/22%	69.6/22%

Table 4: Time domain parameters

	statistical parameters			
	peak (dB)	mean (dB)	s (dB)	RMS (dB)
after one hour	106.9/ 0 %	95.2/ 0 %	91.9/ 0 %	96.3 /0 %
after 4 hours	111.1/ 4 %	97.2/ 2 %	94.5/ 3 %	99.0/ 3 %
after 8 hours	107.6/ 1 %	96.0/1 %	94.6/ 3 %	98.4/ 2 %
after 16 hours	112.4/ 5%	97.8/3 %	96.3/ 5 %	100.2/ 4 %
after 32 hours	108.5/ 2%	96.4/1 %	93.6/ 2 %	98.2/ 2 %
with fault	118.5/11%	104.2/10%	104.5/14%	107.3/11%

Table 5: FWT coefficients parameters .

THRESHOLDS: In order to define thresholds, the maximum dynamic load at meshing have to be compared to a limit value defined by the nature of used materials and the geometry of teeth. The direct measurement of this dynamic load seems to be a delicate operation, thus effect from the dynamic loading can be emphasis either by experimental or numerical means. This step has been limited to the effect of local fault on spur gear. The tested gears are AGMA 12 quality with 36/38 teeth, a contact ratio of 1.3 and a pitch module of 2 mm. The chosen local fault simulates pitch line pitting on a single tooth. The single line of pitting is modelled a a strip of metal removed at the pitch line, a grinding wheel has been used to create this fault. Three levels have been c[illegible]d, their positions were determined using a marking compound (Fig 4).



level N° 1 level N° 2 level N° 3

Fig. 4: Visualisation of different levels of local fault

The geometrical characteristics of local faults for tested gear are presented in Table 6. Unfortunately, it has not been possible to obtain the same thickness and depth of the fault, but the same order of magnitude is obtained.

	Width (l)	Depth (p)	thickness (e)
Level N° 1	4.4 mm	0.043 mm	0.4 mm
Level N° 2	8.7 mm	0.072 mm	0.75 mm
Level N° 3	17.5 mm	0.23 mm	0.97 mm

Table 6: Geometrical characteristics of local fault

The operating conditions were, as usally, five speeds and five applied loads. Only results of RMS values versus speeds for those three levels in time and spectral domains (Fig. 5) will be presented. The acceleration r.m.s. level obtained with and without faults are compared.

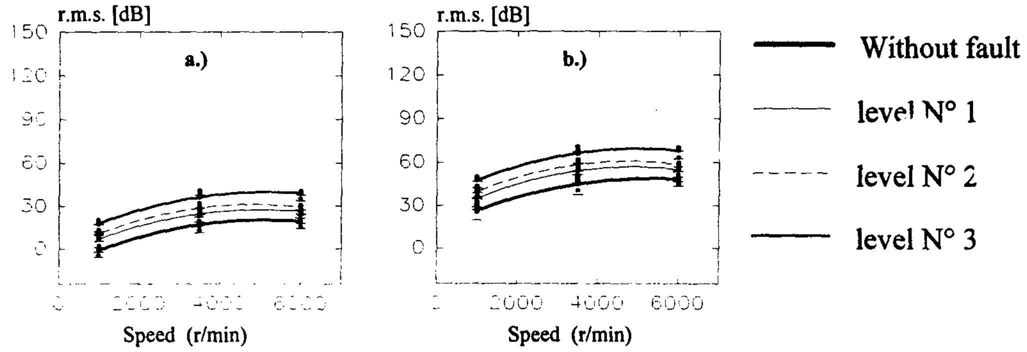


Fig. 5: Effect of local faults on r.m.s. : (a) Time domain, (b) Spectral analysis

An increase in the geometric area of the fault gives an increase of the signal output, in both time and frequency domains. The same trends are observed with respect to torque, but the acceleration r.m.s. increase versus torque is linear. General trends observed for this type of fault have been also mentioned previously in the literature.

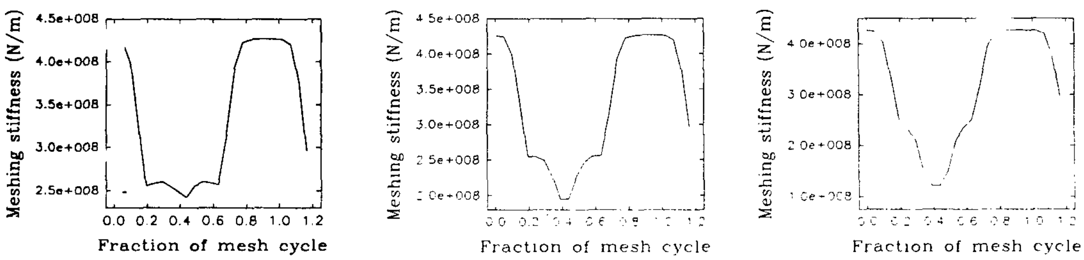
Fitting between experimental and numerical dynamic results is now well known as the global mechanical architecture behaviour is taken into account through realistic bearing stiffness and damping [10]. A mathematical model has been developed to simulate tooth line pitting effect using general equations of motion for torsional response of a geared system and an elastic strain energy calculation of the stiffness function of a gear pair. The Finite Element Method is used to predict the dynamic behaviour of the test rig equiped with faulted gears. The excitations are introduced by the fluctuations of meshing rigidity and static transmission error. These parameters were calculated by taking into account all the design parameters of the gears, using a software based on the Finite Prism Method [11]. This technique allowed a 3D evaluation of cylindrical gear elastic deformations with a fine definition of the load sharing and pressure distribution. For each meshing position, potential areas of gear contact are located, and the unloaded transmission error and relative distances between teeth are calculated and stored. Elastic deformations are then calculated. Local and global deformations are evaluated separately, and surface deformations are estimated by the well known Boussinesq theory. Final results give pressure distributions along

the tooth face width, load sharing between different tooth pairs, stress, meshing stiffness and loaded transmission error. Thus, introducing precise values of meshing stiffness and static transmission errors, finite element modelling of the test rig has been performed. This modelling has been used for the prediction of dynamic effects of gear faults. Shafts were described with single rotor elements with two nodes and six degrees of freedom per node and gears are modelled by rigid disk connected by meshing stiffness. The description of meshing was associated to a 6x6 stiffness (and mass) matrix which connects the 6 degrees of freedom of gear centres. The general form of equations of motions is :

$$M.\ddot{X} + C.\dot{X} + K(\theta^*).X = K(\theta^*).\left\{\xi_o(\theta^*) + \varepsilon_o(\theta^*)\right\} \qquad (1)$$

where θ^* is a space parameter that describes the nominal progression of meshing. It could be expressed by $\theta^* = \Omega^*.t$, where Ω^* is the nominal angular velocity [12]. $\mathbf{K}(\theta^*)$ is the general form of the stiffness matrix (it depends on θ^* due to the meshing stiffness variations) and $\xi_0(\theta^*)+\varepsilon_0(\theta^*)$ is a specific vector which simulates excitation associated with the static transmission error variations and the unloaded transmission error[13]. M and C are mass and damping matrix. This system of equations is solved using the step by step Newmark techniques. A preliminary analysis begins by a geometrical description of the gear teeth, including profile modifications, manufacturing errors, gear bodies position including eccentricity, centre distance faults and misalignments. Then, gear meshing progress is analysed during a large angular period. This rigidity was then applied on the dynamic model and the tests rig was modelled by 41 elements with 251 degrees of freedom. The chosen modal damping was 0.02 for all modes of the stucture and 0.06 for mesh.

RESULTS: Local gear faults, described in the previous experimental part, have been introduced in this analysis by their geometrical description on a single tooth flank. The effects on meshing stiffness are presented in the figure 6. The results show that fault levels N°1 and N°2 have practically no significant effect on meshing stiffness variation, but level N°3 gives a variation of 50%. The geometrical variation due to gear fault can lead to a modification of the kinematic motion, a pseudo-polygonal effect appears due to flat bearing segment along the tooth profile caused by the fault geometry.



level N°1 level N°2 level N°3

Fig. 6: Effects of faults on meshing stiffness evolution

The static transmission error has been determined using a kinematic simulation of mesh for fault level 3 only (Fig. 7) as for the others levels, this excitation is considered negligible.

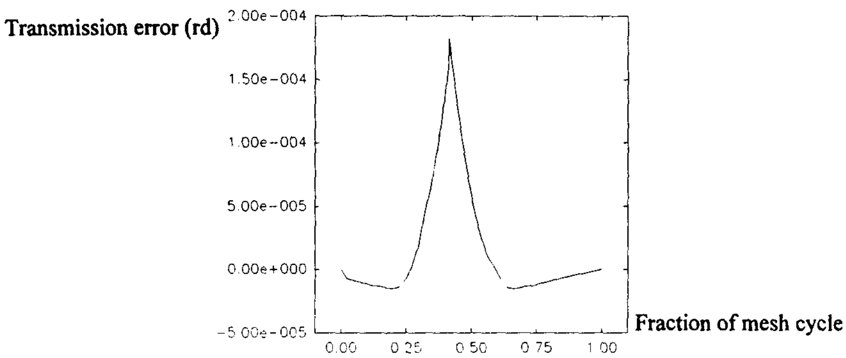


Fig. 7: Transmission error-fault level N°3

Calculations have been performed for different values of torque: the acceleration r.m.s. values of the spectral analysis have a linear variation with respect to torque, so, the r.m.s. values are calculated for several speeds and the maximum value of torque (120 Nm), in a steady state regime at each step of speed (50 r/min) (Fig.8).

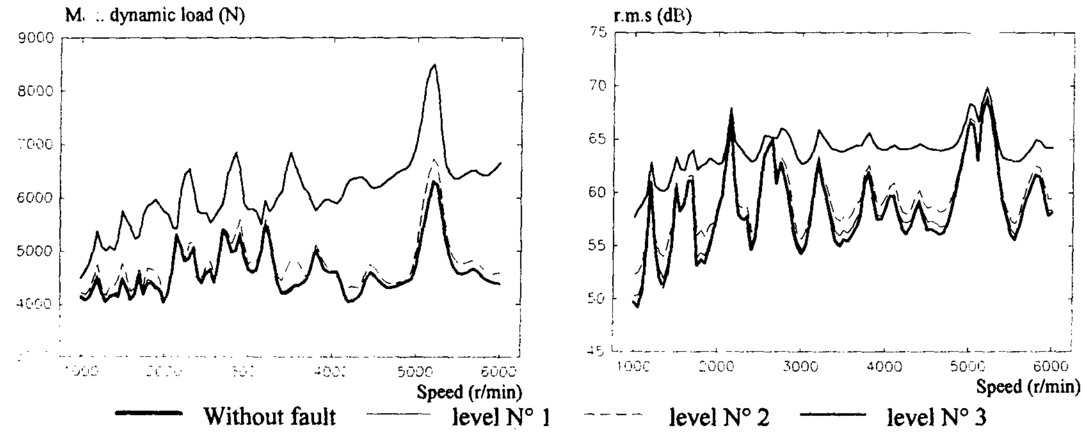


Fig. 8 : RMS acceleration values & Max. dynamic load

The maximum increase of the acceleration RMS is about 10 dB, while it was about 22 dB from measurements. Nevertheless, general trends are almost the same. We can observe that the increase of RMS values is almost negligible around pics, this is due to the fact that the effects of tooth deflection at those speeds are more important than the modification of tooth mesh rigidity or static transmission error due to fault.

CONCLUSION: Three topics have been considered in this work, the influence of design parameters on the dynamic behaviour of gearbox, which enable users to optimize gear's design. Then the influence of local and uniformely distributed faults have been studied, this part permit to establish a data base on the dynamic behaviour of gears with faults. Discriptors have been defined in order to indicate early the presence of failure. The last part deals with the elaboration of numerical model in order to establish thresholds. The same general trends have been observed

in both experiments and calculations. The next step is to determine the maximum stress due to this maximum dynamic load and to make comparaison with respect to the limit value for rupture. The calculations were not performed yet, What we are looking for now is to adjust the numerical model and particularly the simulation of excitation forces and the adjusting of different parameters as damping and static transmission error. Experimental work have been done to study the response of gearbox with faults on bearing, in the same time we are working to elaborate a numerical model to take into account the dynamic behaviour of bearing as a rolling bodies, the aim is to study gearbox dynamic behaviour with faults on both gears and bearing.

REFERENCES

[1]-ALATTASS M., Maintenance des machines tournantes: signature de defauts d'engrenage droits et helicoidaux, Thèse de doctorat, I.N.S.A de Lyon, 1994, 198 p.

[2]-RANDALL R.B., A new method of modelling gear faults, Jou⁻al of ..chanical design, ASME, 1982, 104, pp 259-267.

[3]-ESHLEMAN R.L., The role of sum and difference frequencies in rotating machinery fault diagnosis, 2nd international conference on vibration in rotating machines, Cambridge, 1980, pp 145-149.

[4]-TAYLOR J.L., Fault diagnosis of gears using spectrum analysis, 2nd international conference on vibration in rotating machines, Cambridge, 1980, pp 163-168.

[5]-ALATTASS M., MAHFOUDH J. & PLAY D., Morphological study of vibratory signals, Conference on the acoustical and vibratory surveillance methods and diagnostic techniques, Paris, 1995, pp 456-474, CETIM.

[6]-MAHFOUDH J., BARD C., ALATTASS M. & PLAY D., Simulation of gearbox dynamic behaviour with gear faults, Second International Conference on Gearbox Noise, Vibration &Diagnostics, LONDON, 1995, pp.91-100, IMECH E.

[7]-BARD C., REMOND D. & PLAY D., New transmission error measurement for heavy load gears, Proc. of the 1994 International Gearing Conference, 1994, University of Newcastle upon Tyne, UK, pp.393-399.

[8]-Remond, D. , Bard, C., and Play, D., A Systematic Approach of the effect of Design Parameters on Spectral and Wavelet Gear Noise Analysis, JSME International Conference on motion and power transmission- Hiroshima, Japan- pp 128-1?? (Nov. 1991)

[9]-Fauchon, J., "Plans d'Experiences" pp 99, INSA- LYON (1991). cours polycopié du département de génie mécanique développement.

[10]-BARD C., Modélisation du comportement dynamique des transmission par engrenages, Thèse de Doctorat, INSA de LYON, 1995, 292 p.

[11]-OLAKOREDE A., PLAY D., Load sharing load distribution and stress analysis of cylindrical gears by Finite Prism Method in a CAD environment, Proc. Design Productivity Int. Conf., Feb. 1991, Honolulu, Ragsdell & T.Holt, pp 921-927.

[12]-BLANKENSHIP G. W. & SINGH R., A comparative study of selected gear mesh interface dynamic models, International Power Transmission and Gearing Conference, A.S.M.E, 1992, Vol. 1 pp 137-146.

[13]-ÖZGUVEN N. & HOUSER D.R., Mathematical models used in gear dynamics - a review, Journal of Sound and Vibration, 1988, N° 121(3) pp 383-411.